MECHANICAL ENGINEERING DEPARTMENT

SCHOOL OF ENGINEERING

THE UNIVERSITY OF CONNECTICUT

INTERIM PROGRESS REPORT

for the period

November 1, 1964 through April 30, 1965

on

HEAT TRANSFER STUDIES OF VAPOR CONDENSING AT HIGH VELOCITIES IN STRAIGHT TUBES

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Submitted to

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

(Code SC-NsG-204-62-S1)

Storrs, Connecticut

June, 1965

Introduction

This Interim Progress Report covers research done under Contract No. NASA Code SC-NsG-204-62-S1 for the period November 1, 1964 through April 30, 1965.

This is the third semi-annual interim report submitted during the period of the current grant. The two previous reports were submitted on the respective dates of May 1, 1964 and November 2, 1964.

The current report is concerned primarily with the following areas of research:

1. Experimental measurement of the flow dynamics of the annular liquid layer
2. Determination of the amount of liquid entrainment in the vapor core in annular two-phase flow
3. Reorganization of computer programming of data reduction process
4. Reorganization of and improvement of computer programming of solutions of annular flow mathematical model
5. Construction of instrumented one-inch diameter condenser with improved instrumentation for additional heat transfer measurements in condensing flow
6. Derivation of a mathematical model for annular-mist flow

Abstract

This report includes short descriptions of the progress of current work in the several areas of research listed on the previous page in the Introduction.

Experimental apparatus has been successfully developed for measuring the flow characteristics in annular wave motion. Some experimental data on the velocity, frequency and height of annular waves is reported.

From experimental impact pressure measurements with a 'Dussourd' type impact probe, calculated data is presented, graphically showing the concentration variation of liquid particles in the vapor core.

A print-out statement of the computer program for data reduction is included and the progress of the computer solution program for the mathematical model is discussed.

A discussion and diagram of the condenser test unit under construction is given. This test unit is similar in dimensions to the model now in use, however, it is instrumented with additional and improved thermocouple installations so as to permit more precise determination of the local condensing heat transfer coefficient on the inside tube surface.

A description of the 'Annular Mist Flow' model is given, along with a discussion of the development of the mathematical analysis for this system which has been undertaken.

Liquid Layer Measurements

The test section described on page 1 of the second Interim Report of November 2, 1964 for measurement of liquid layer properties has been redesigned and in operation for several months. Problems involving the circuitry in the counting device have been solved and a method for measuring the velocity of the waves on the liquid layer has been devised. Recent results show good repeatability helping to establish a high level of confidence in the data.

Briefly, the probes have been changed from those described in the earlier report in the following manner: the probe is now mounted in such a manner that the turning of a thumbscrew moves the probe up or down, the amount of movement is shown by a dial indicator reading in thousandths of an inch. This drive mechanism is now thermally insulated from the test section. The use of the dial indicator has proven to give a more precise probe position indication than the direct micrometer drive mechanism used previously.

The theory of operation of the probes remains the same as previously detailed in the second Interim Report of November 2, 1964, but will be reviewed briefly here. There are two needle-contact probes mounted on each drive unit. The drive units are located fore and aft of a short condenser section. The probes enter the tube through seals in the tube wall and extend over the tube cross section into the liquid layer on the opposite wall. The probes are supplied with a positive d.c. voltage and the tube wall is at ground. When the probes contact the liquid film on the tube wall the circuit is completed and a signal emanates from each probe. The absence of a signal means the probe is in the steam core. The signals are led both to a decade counter and to a dual beam oscilloscope. The counter uses only the signal from the first of the two contact needles, which is located 0.500 ± 0.005 inches from the rear needle, giving the number of voltage changes over a certain counting period (10 sec.). Each of these voltage changes indicates contact with a liquid wave. The oscilloscope displays the signals of both probes. There is a characteristic scope pattern for the probes in the steam core and a different pattern for the probes in the liquid layer; when in contact with the waves a combination of the two occurs. By comparing photographs of these patterns one can determine the time it takes a particular signal (i.e., a particular wave) to travel from the first to the second probe. This is, of course, the wave velocity. This method works quite well for deep waves but for shallow waves it has proven unusable since there is some chopping of the wave crest. This chopping effect is probably of the same order of magnitude as the dimension of the small wave. At the present time, these are the only known direct measurements of wave speeds in condensing flow.

The range of flow parameters covered at this time is not extensive. The vapor velocities are in the range of 200-400 fps and total mass qualities are high (greater than 80%). Within the above limitations of flow conditions the following results occurred:

1. With 10% to 20% of the total flow condensed between test probe one and test probe two (approximately two feet) the average film thickness as a function of angular position changes from annular at probe I (average thickness of approximately 0.002 inches) to nearly stratified flow at probe II (the average

thickness of the bottom is approximately 0.046 inches decreasing rapidly to approximately 0.020 inches at 45° up from the tube bottom, then slowly falling to approximately 0.007 inches at the top of the tube).

2. The ratio of maximum wave height to minimum wave height (thickness of undisturbed liquid layer) generally decreases from the bottom to the top of the pipe. The value of this ratio seems to be a function of quality, the thicker liquid layers having the largest numerical value of the ratio.

3. The wave velocities near the crest of the wave are in the order of 10-12 fps and those at the wave trough are approximately 3-4 fps. These relative velocities between crest and trough indicate the presence of roll waves.

Figure 2 shows a plot of counts per second versus distance from the wall. To establish wave dimensions an arbitrary cutoff point was established, i.e., the maximum and minimum wave heights are at those points where the curve intersects the two counts per second line. The average wave height is determined by the intersection of a horizontal tangent with curve.

Figure 3 shows the peripheral distribution of the average liquid layer thickness. It shows the nearly symmetric thickness distribution at high qualities (Station I) and the stratification effects at lower qualities.

Figure 4 is a reproduction of a oscilloscope photo showing the method of determining wave speeds.

Experimental Determination of the Quantity of Liquid Particle Entrainment in The Vapor Core in Annular Two-Phase Flow

The mathematical model used to describe annular condensing flow used throughout this investigation has been based on the assumption of a pure saturated vapor core. It was known from the beginning that this assumption was inexact. However, no useful quantitative data was available as to the actual liquid droplet content to be expected in the vapor core of an annular flow condensing system.

As reported on page 5 of the Interim Progress Report of May 1, 1964, a so-called 'Dussourd' type impact probe has been constructed (see also Ref. 4) which makes possible the measurement of the true vapor velocity in a liquid droplet laden vapor stream. A considerable amount of experimental information has been recorded during this investigation, which was made possible by use of this instrument.

Following is an analysis of one aspect of the 'Dussourd' probe data, which makes possible the experimental determination of the liquid droplet concentration in the vapor core of the condensing system.

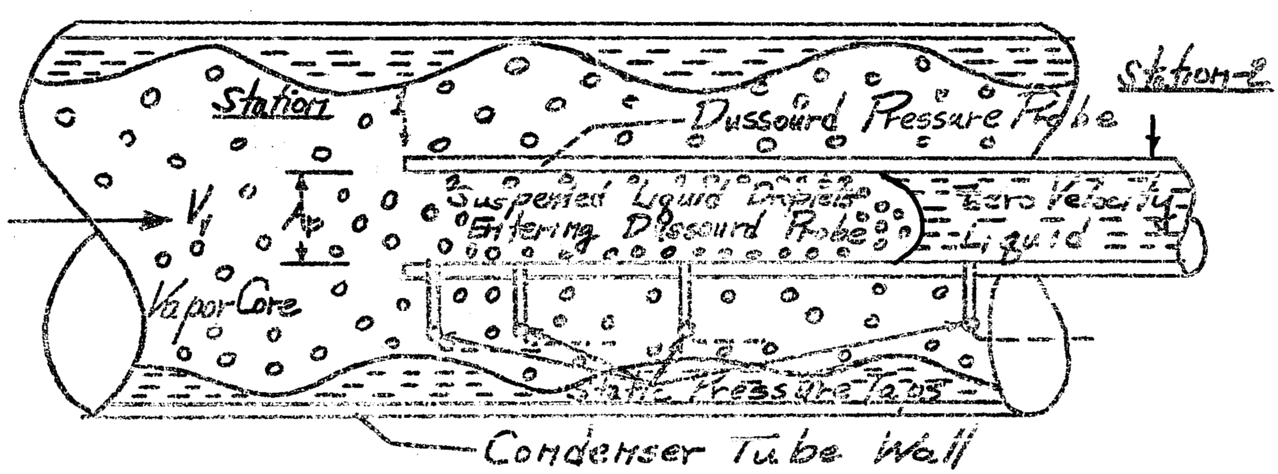


Figure 1

Referring to Figure 1, the following assumptions are necessary in order to calculate from the experimental data the droplet concentration in the vapor core of the annular system.

1. The droplet velocity entering the probe mouth is the same as the vapor velocity of the local free stream.
2. The capture efficiency of the liquid droplets by the Dussourd probe from the free stream is essentially 100 percent over the cross section of the probe mouth.
3. The static pressure difference between Stations (1) and (2) along the probe represents the so-called error in the measurement of the impact pressure of the vapor by use of the pressure at Station (2) as the stagnation pressure of the vapor. Thus for the vapor

$$p_0 = p_2 - (p_2 - p_1) \tag{1}$$

or

$$p_0 = p_2 - \Delta p_p \tag{2}$$

where $\Delta p = p_2 - p_1$ is determined by extrapolating the experimental measurements taken at the static pressure taps to the mouth of the probe.

Considering the momentum interchange that takes place between the mouth of the probe and the liquid interface inside the probe, one may write

$$\Delta p_p A_p = \mathring{m}_{Lp} V_{Lp} \tag{3}$$

where $\dot{m}_{Lp}$ is the total mass flow rate of liquid droplets entering the mouth of the probe at a velocity V_{Lp}.

One may now define the local vapor concentration (quality) at the probe mouth by

$$x \equiv \frac{\dot{m}_{vp}}{\dot{m}_{vp} + \dot{m}_{Lp}} \tag{4}$$

where $\dot{m}_{vp}$ is the mass flow rate of saturated vapor in the free stream at this location for a cross section equal to that of the tube mouth.

Equation (4) may be written

$$x = \frac{1}{1 + \dot{m}_{Lp}/\dot{m}_{vp}} \tag{5}$$

From the principle of continuity of mass flow in this steady state system, we may write

$$\dot{m}_{vp} = V_v \, \bar{A}_v \, \rho_v \tag{6}$$

and

$$\dot{m}_{Lp} = V_v \, \bar{A}_L \, \rho_L \tag{7}$$

where $\bar{A}_v$ and $\bar{A}_L$ are the apparent cross section areas occupied by the respective phases at the probe mouth, and

$$A_p = \bar{A}_v + \bar{A}_L \tag{8}$$

where A_p is the inside cross section area of the probe mouth.

Substituting from Equations (6) and (7) in Equation (5) one obtains

$$x = \frac{1}{1 + \dfrac{\bar{A}_L \, \rho_L}{\bar{A}_v \, \rho_v}} \tag{9}$$

Now from Equations (3), (6), (7), and (8) we may substitute for the cross section area terms A_L, A_v, and A_p in terms of the known or measured quantities ρ_v, ρ_L, and p_p. We then find that the expression for droplet concentration becomes

$$x = \frac{1}{1 + \left(\frac{1}{V_{Lp}^2 \; \rho_L/\Delta p - 1} \right) \left(\frac{\rho_L}{\rho_v} \right)} \tag{10}$$

It is worthwhile to examine Equation (10) for the limiting conditions of (a) pure vapor flow (zero droplet concentration) and (b) maximum liquid droplet concentration. For case (a) with very few liquid particles present $\Delta p \to 0$ and Equation (10) approaches

$$x \to \frac{1}{1 + \frac{1}{\infty} \left(\frac{\rho_L}{\rho_v} \right)} = 1 \tag{11}$$

If the probe mouth were filled with discrete liquid particles, the maximum possible concentration of liquid would exist, case (b). If it is assumed that the liquid particles nearly fill the probe mouth, then $\overline{A}_{Lp} \approx A_p$ and we may write

$$\dot{m}_{Lp} = A_p \; V_{Lp} \; \rho_L \tag{12}$$

Then from Equation (3) we find

$$\Delta p_p \; A_p = A_p \; V_{Lp}^2 \; \rho_L \tag{13}$$

which reduces to

$$\frac{V_{Lp}^2 \; \rho_L}{\Delta p_p} = 1 \tag{14}$$

Substituting Equation (14) into Equation (10) gives

$$x = \frac{1}{1 + (\frac{1}{1-1})\frac{\rho_L}{\rho_v}} \tag{15}$$

which reduces to

$$x = \frac{1}{1 + \infty} = 0 \tag{16}$$

indicating that no vapor is present as was assumed at the beginning of this paragraph.

Thus Equation (10) satisfies both limiting conditions of maximum and minimum droplet concentrations.

We see, therefore, that within the limitations of the three necessary assumptions described at the beginning of this section, Equation (10) can be used to calculate the droplet concentration in the vapor core of an annular condensing stream.

The distribution of droplet concentration in the core of a condensing annular flow has been calculated from the experimental probe data for a range of test data covering total mass flow conditions from a pure vapor flow to very low vapor content, with a vapor velocity range in excess of 800 feet per second. Figures (5 through 7) show the variation in droplet concentration by percent of mass flow for a number of different tests. These figures clearly show that the droplet mass concentration in the vapor remains below 10% over the major portion of the condensing tube. In addition, the droplet concentration seems to be primarily a function of the mass flow ratio of vapor to liquid in the tube.

It is expected that additional Dussourd probe measurements will be made in the future to further confirm the results of these experiments. It is also expected that the information developed to date will be useful in the development of the annular mist flow mathematical model discussed elsewhere in this report.

Computer Program for the Reduction of Experimental Data In Annular Two-Phase Flow

In the first Interim Progress Report of May 1, 1965 the system of experimental data reduction was discussed on pages 7 through 10 of that report. Very briefly, the system consists of several preliminary calculations and tabulation of the experimental data which is then punched onto I.B.M. cards. The card deck is then processed by an I.B.M. 7040 program; a copy of the print-out statement of one such program is given in Figure 8 of this report. This data reduction program is under continual review and modification as changes are made in either the experimental apparatus, the experimental operating conditions, and/or the theoretical model used in the analysis of the system. An example of one change was a modification which made possible the acceptance of inlet conditions which could include saturated liquid along with the saturated vapor. Another modification was the addition of a subroutine which will determine the local condensing heat transfer coefficients along the tube axis.

A print-out statement of the complete data reduction program will be included in the Final Report.

Computer Program for the Analytical Solution of the System of Equations Describing the Mathematical Model

One fundamental objective of this research project was the development of a system of equations whose solution would satisfactorily represent the local flow properties found in a high capacity condensing tube. A system of such equations as well as several solutions of this system were reported in the Final Report on Heat Transfer Studies of Vapor Condensing at High Velocities in Small Straight Tubes for the period November 1, 1961 through October 31, 1963 (see Ref. 1). The computer solution of this same system of equations has been converted to run on the I.B.M. 7040 digital computer. Several changes and modifications are incorporated into this new computer program. They are as follows: (1) the use of a steam table subroutine in place of the present exponential method to predict thermodynamic properties, (2) the introduction of higher order methods in the numerical integration process, and (3) a polynomial curve fitting subroutine to assist in predicting the derivatives of the changing thermodynamic properties of the fluid.

Construction of a One-Inch I.D. Condenser Tube with Improved Instrumentation For Making Additional Heat Transfer Measurements

The instrumentation and construction details of a one-inch I.D. 20-foot long condenser test rig are described in the first Interim Progress Report of May 1, 1964 for this project. The results of a series of condensing tests made using this test unit were reported in the second Interim Progress Report of November 2, 1964. This same test unit has been in constant use during the period of the current report. The current experimental work is concerned primarily with the determination of condensing flow dynamics, such as wave height, speed, and frequency. This test work, which is described elsewhere in this report, is still in progress.

Experience with this one-inch I.D. test condenser, has revealed some shortcomings in instrumentation and construction.

The problem of the measurement of the local outside surface temperature of the condenser tube, has given the most difficulty with respect to dependability and precision of measurement on the current test unit. These surface temperature measurements are vital to the experimental determination of the inside condensing surface coefficient of heat transfer.

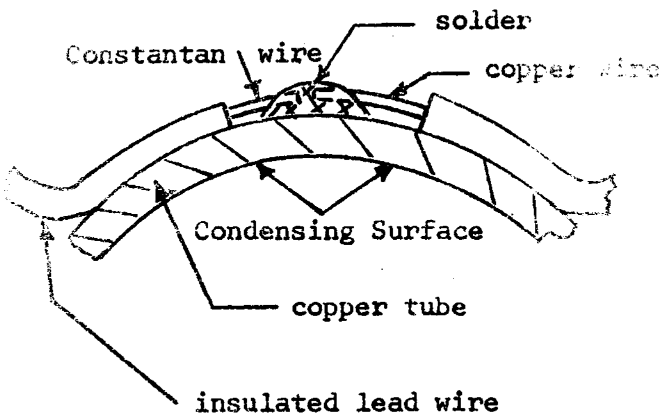
In order to make additional more accurate condensing heat transfer measurements a new condenser tube is being constructed. The basic design of the new test unit is essentially the same as the one currently in use, however, certain changes and improvements have been incorporated. Most of these changes are detailed in Figures 9 and 10. Probably the most important change in construction involves doubling the number of thermocouples used to measure both the outside surface temperature of the condenser tube and the local temperature of the cooling water in the annulus. Other changes include an enlarged inlet plenum chamber and lengthened adiabatic inlet section of the tube itself. Local static pressure taps are attached to the outside wall of the condensing tube with silver solder instead of lead solder. For the purpose of visual observation and photography, a transparent glass section will be installed downstream from the contact wave probe apparatus now in use. A much smaller diameter traveling impact probe will be used along with the so-called Dussourd type probe now in use. It appears that the Dussourd total pressure correction measurements already taken can be calibrated against total vapor to liquid flow ratio and used in conjunction with a much smaller conventional impact probe for most tests. This will considerably reduce the problem of flow variation and adjustments as well as variation in pressure losses in the tube due to the presence and position of the large Dussourd type impact probe.

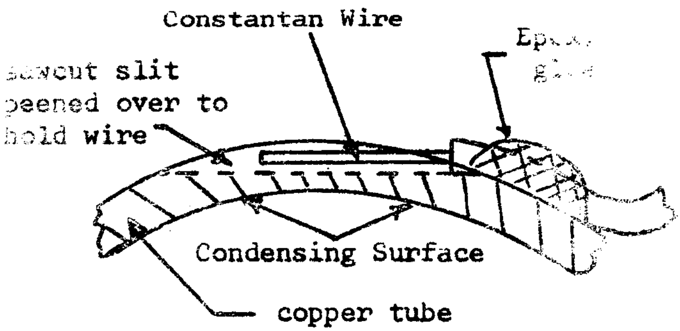
Additional discussion of the problem of surface temperature measurements on the O.D. of the condenser tube is warranted at this point. The technique of attaching a thermocouple satisfactorily to the outer surface of the condenser tube is a delicate task for this installation. The heat transfer system under examination is one in which the major portion of the total resistance (maximum temperature drop) to radial heat transfer from the condensing vapor occurs in the comparatively thin liquid layers on both sides of the tube wall. The resistance of the copper wall is not a major factor in the total resistance and in any case can be calculated with acceptable accuracy. The resistance of the liquid layer, of the cooling fluid on the outside wall of the condenser tube, is usually greater than that in the liquid layer on the condensing side. This means that the temperature drop in the cooling fluid film is the major portion of the radial temperature difference between the two fluids. Any object such as a thermocouple attached to the outside surface is bound to affect the temperature distribution in this outer film to some degree as well as the surface temperature at the point of attachment. A further complication of the problem is the tendency to establish a small node or lump on the surface, with whatever method of attachment is devised. The situation sometimes arises that the thermocouple records the temperature at some location in the lump rather than the temperature of the undisturbed surface. Any lump or node tends to act to some degree as extended surface with a temperature somewhat less than that of the original surface, provided of course that the thermal conductivity of the material in the lump is not significantly less than the conductivity of the tube wall. This then is another example of the classic problem of the instrumentation affecting the measurement it is intended to measure.

The design of the thermocouple installation to be used in this new test unit is intended to circumvent these problems as nearly as possible. The thermocouple joint is to be submerged just below the surface of the tube wall with the outer tube surface peened over the thermocouple lead wires and smoothed to conform as nearly as possible to the original surface. The two lead wires then follow the submerged channel circumferentially a short distance in opposite directions from the joint. They then emerge from the surface and follow close to the tube surface circumferentially a quarter of the way around the tube until each wire meets a wire from the diametrically opposite thermocouple. These two unconnected wires are then led out of the annulus in an axial direction in the same sheath (see Figure 10). Outside the test unit, the proper wires are then recombined at the thermocouple junction switch. The diametrically opposite thermocouples on the outside tube surface are intended to facilitate the discovery of a bad thermocouple installation as well as to improve overall accuracy by averaging any circumferential temperature variation due to circumferential variations in fluid flow distribution either inside or outside the condenser tube. Some experimental work has been done to determine a better thermo-mechanical means of installing the surface thermocouples so as to give the best results. Table 1 summarizes some of the details considered in this work. The problem of locating and securing the thermocouple lead wires and static pressure lead lines in the cooling annulus has been considered. The method selected was the installation of a series of circumferential rings slightly smaller in diameter than the I.D. of the outer tube. The pressure lead lines and thermocouple wires are to be secured to the circumferential rings and held as closely as possible to the inside surface of the outer tube.

With the new test unit it is expected that the outer surface temperature of the condenser tube can be determined with sufficient accuracy. This should make possible the determination of the local inside surface coefficient along the length of the one-inch I.D. condensing tube. The resulting test data will then be compared with similar data recorded previously on smaller condenser tubes, both at the University of Connecticut and at the Lewis Laboratory.

Table [illegible] Experience with Previous Thermocouple Installations

<table>
<tr><th>Schematic (Enlarged)</th><th>Description</th><th>Comment</th></tr>
<tr><td>
</td><td>Two wires soldered on to the outer surface of the tube.

Variation: Silver solder or braze used instead of soft solder.</td><td>1. Creation of a flat lump of solder on the surface was unavoidable in order to attach the wires satisfactorily to the surface.
2. Since a major portion of the total temperature drop occurs near the tube wall on the coolant side the protrusion of the lump significantly affects the local wall temperature causing temperature variations due to installation differences.
3. Use of silver solder at this connection damages (melted) thermocouple insulation.</td></tr>
<tr><td>
</td><td>One constantan wire is placed in a circumferential slit in the tube surface and the copper is peened over the wire with a chisel to hold the wire. The insulated wire is attached with epoxy to the tube wall.
Variation: A copper wire was similarly attached 1/2 inch downstream to make a two-wire junction to the tube.</td><td rowspan="2">In this installation the copper tube was used as one leg of the thermocouple. This system seemed to be most satisfactory from a practical viewpoint and exhibited the lowest temperature variations (10° max.). However, the attachment of the insulated wire epoxied to the tube wall was ineffective. Failure of this attachment introduces thermal conduction errors which can only be estimated. The axial temperature gradient (150°max.) along the tube introduces thermocouple calibration error due to variation in the thermo-electric property of the tube with temperature. The use of the two-wire system did not seem to reduce the resulting surface temperature variations and unfortunately it increased the incidence of joint failures at the tube-wire joint.</td></tr>
<tr><td colspan="2">Other Methods:
Several methods of attachment to the tube wall were tried such as drilling holes or chiseling up a flap of metal and then placing the thermocouple leads in the opening and peening the copper to hold the wires. These operations sometimes penetrated the tube wall and were, therefore, unsatisfactory.
Copper and constantan ther[illegible] leads were soft soldered, fused or brazed and placed in the sawcut [illegible] the copper tube (see above) which was then peened over. These joi[illegible] not necessary to form a thermocouple in this case and proved [illegible]ysically large and often broke during the peening operation.</td></tr>
</table>

Two-Phase, Annular-Dispersed Flow Model

(Annular Mist Flow)

Two-phase, annular-dispersed flow may be defined as a continuous gas phase containing liquid particles (droplets) surrounded by a liquid annulus in contact with the solid boundary of the duct. During the current phase of the work, one of the limits of this model, that of pure annular flow (i.e., no entrained liquid particles in the gas phase) has been analytically investigated. Inasmuch as the present experimental work with the Dussourd probe has definitely shown the existence of liquid particles in the gaseous core, the extension of the pure annular flow model to an annular-dispersed flow model is a natural step in the analytical investigation of two-phase, single-component, condensing flows.

As in the case of the current annular mathematical model the annular mist model is one dimensional and assumes quasi-thermodynamic equilibrium (i.e., equality of temperature, chemical potential, and pressure, but not of velocity) between the continuous gas phase and liquid annulus. Thus, the principle difference between the two mathematical models will be the inclusion of the interaction between the liquid particles and the gas phase. Here, a similar quasi-thermodynamic equilibrium is assumed to exist.

At this point, a discussion of the above assumptions seems warranted. The assumption of an equality of temperature across any cross section of the tube implies an infinite heat transfer coefficient between the gas phase and the liquid annulus, with which it is in contact--just as an equal interfacial velocity would imply an infinite drag coefficient at the interface. This assumption may be too restrictive, and the elimination of it would perhaps be a further step in the improvement of the one-dimensional model. The assumption of an equality of chemical potential does not enter explicitly into the gas-liquid annulus interaction, however, the equality of pressure and temperature would imply this. For the gas-liquid particle interaction, the equality of chemical potential could not be assumed if nucleation theory is to be brought into the model.

The assumption of an equality of pressure between the gas and liquid annulus follows from experimental evidence. For the gas-liquid particles this assumption is quite reasonable if the particle sizes are not too small, as seen by inspection of the Kelvin-Helmholtz equation which states that the difference between the pressure of a spherical liquid droplet and the equilibrium vapor pressure at the gas temperature is inversely proportional to the droplet radius.

It is believed that as in the case of the current model a system of ordinary, first order, non-linear simultaneous differential equations can describe the annular-mist model. The system must contain the six dependent variables (pressure, vapor mass flow rate, liquid particle mass flow rate, vapor velocity, liquid annulus velocity, liquid particle velocity) and the one independent variable--axial distance along the duct. The governing equations for this system are the combined momentum, vapor momentum, liquid particle momentum and combined energy equations. One problem which arises is the prediction of the liquid particle growth and size. This difficulty may be attacked in either an analytical or empirical manner. A semi-analytical approach requires use of nucleation theory to predict particle concentration and growth along with a critical Weber number approach to predict the size of particles which are sheared from the wavelike liquid annulus-gas interface. An empirical treatment would make use of the apparently reproducible data from the Dussourd probe to predict particle concentration, and a critical Weber number to predict particle size.

Such a system of non-linear differential equations may be programmed for solution by a numerical integration process on an I.B.M. 7040 digital computer, as is done with the current model. To investigate the stability of the solution, several integration schemes of various order can be used, and to further increase the accuracy, subroutines predicting saturated liquid and vapor properties, and slopes of these properties to a high degree of accuracy may be utilized.

References

1. W. E. Hilding et al., "Final Report on an Experimental and Analytical Study of the Heat Transfer and Pressure Drop of Vapor Condensing at High Velocities in Small Straight Tubes," October 1963.

2. C. H. Coogan, Jr., W. E. Hilding et al., "Interim Progress Report on Heat Transfer Studies of Vapor Condensing at High Velocities in Straight Tubes," May 1964.

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4. J. L. Dussourd, "A Theoretical and Experimental Investigation of a Deceleration Probe for Measurement of Several Properties of a Droplet Laden Stream," Research Report MIT Sc.D. Thesis. Oct. 1954.

5. Steltz and Silvestri, "The Formulation of Steam Properties for Digital Computer Application," ASME Transactions, vol. 80, No. 4, pp. 967-973, May 1958.

6. SHARE #1095 "WH58, Thermodynamic Properties of Steam and Water."

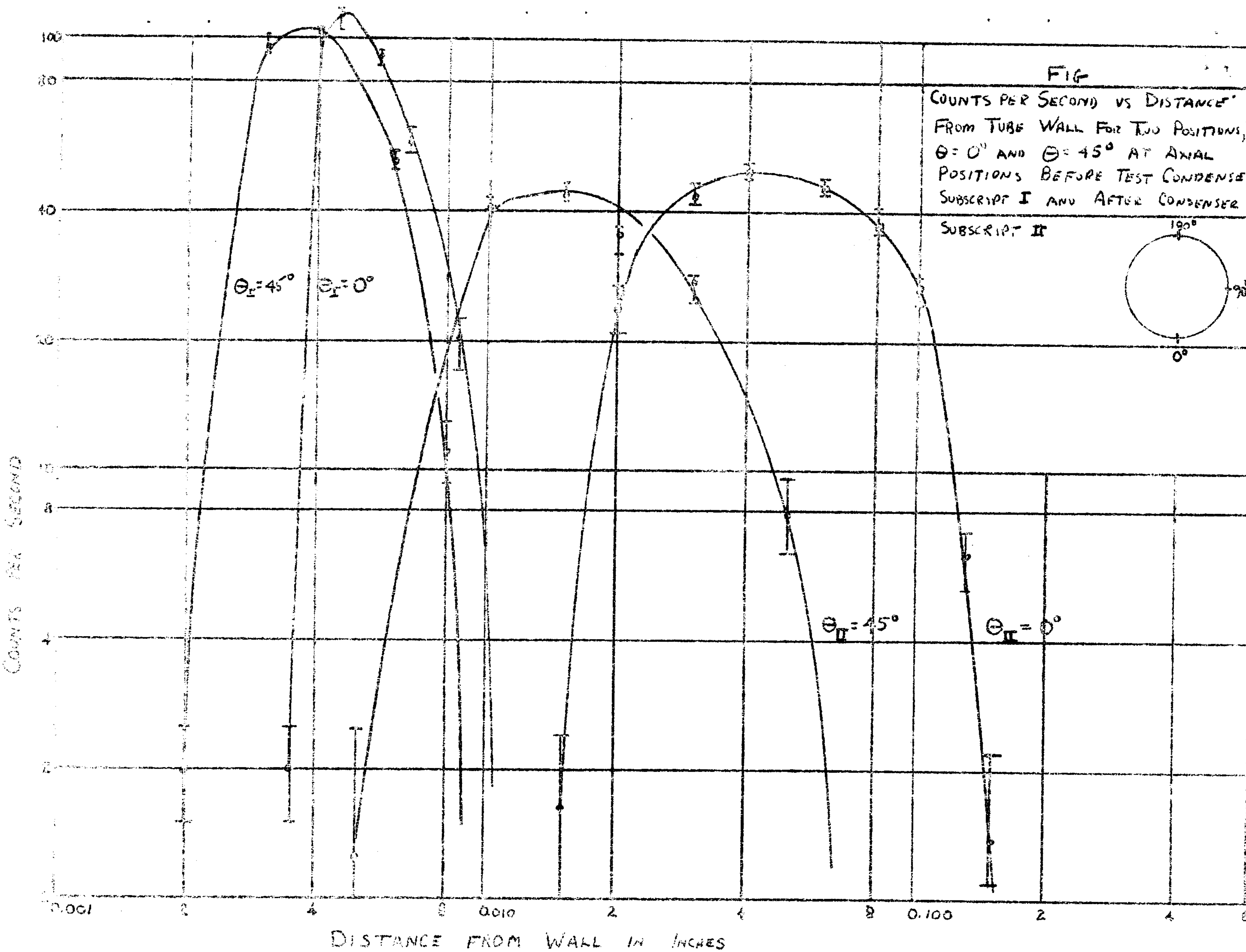
FIG
COUNTS PER SECOND VS DISTANCE FROM TUBE WALL FOR TWO POSITIONS; θ = 0° AND θ = 45° AT AXIAL POSITIONS BEFORE TEST CONDENSER SUBSCRIPT I AND AFTER CONDENSER SUBSCRIPT II
180°
90°
0°
θ_I = 45°
θ_I = 0°
θ_II = 45°
θ_II = 0°
100
80
40
20
10
8
4
2
1
0.001
2
4
8
0.010
2
4
8
0.100
2
4
6
COUNTS PER SECOND
DISTANCE FROM WALL IN INCHES

Figure 2

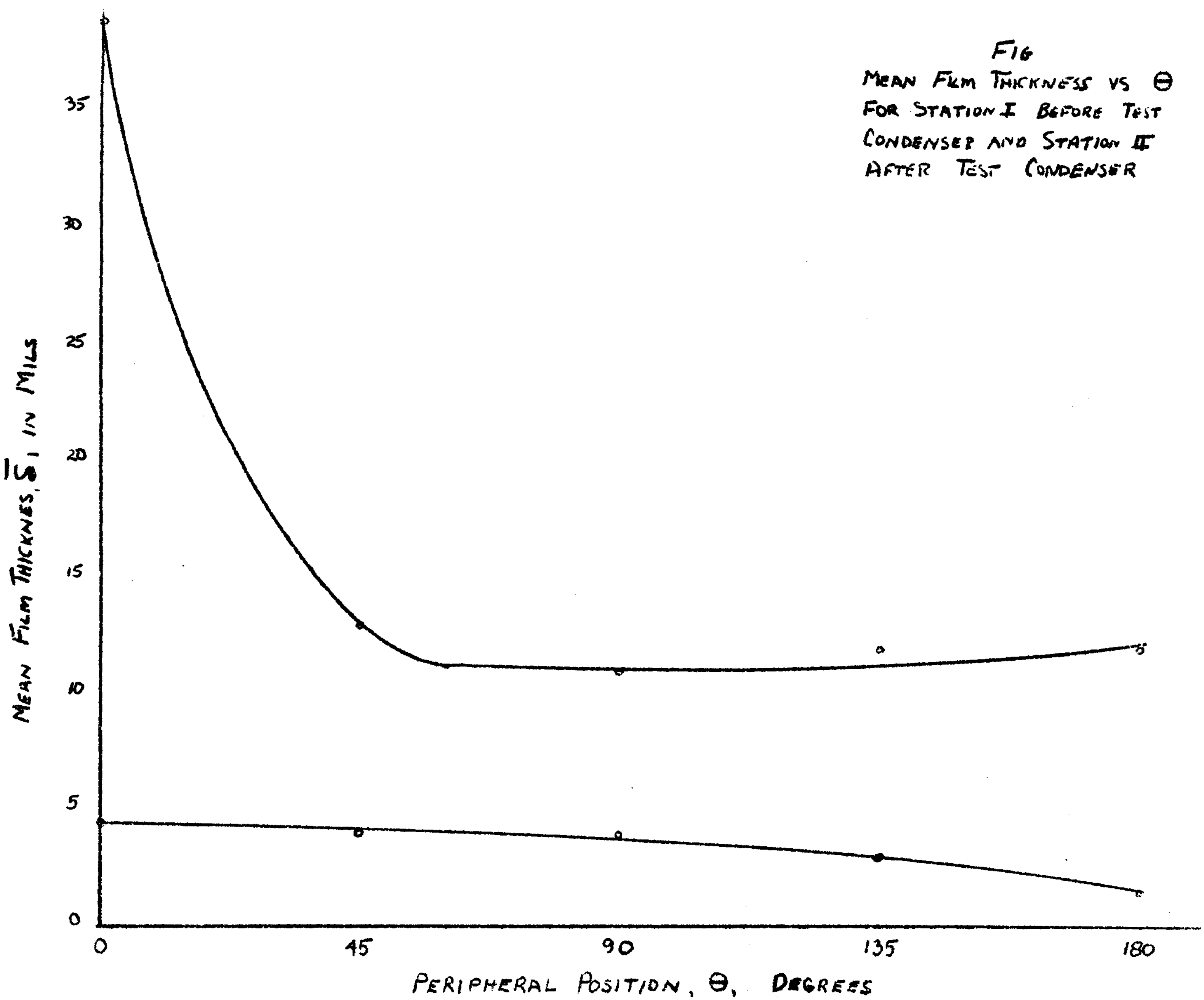
FIG
MEAN FILM THICKNESS VS Θ
FOR STATION I BEFORE TEST
CONDENSER AND STATION II
AFTER TEST CONDENSER
MEAN FILM THICKNES, $\bar{\delta}_1$, IN MILS
35
30
25
20
15
10
5
0
0
45
90
135
180
PERIPHERAL POSITION, Θ, DEGREES

Figure 3

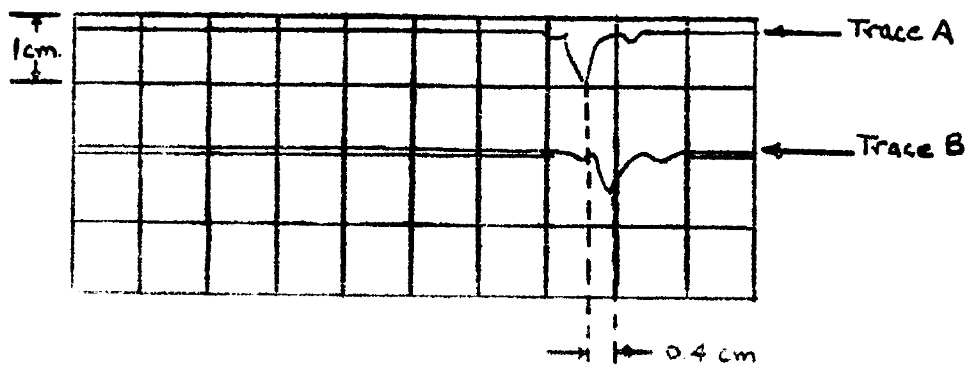

Figure Oscilloscope Photo, Test 62965.

The 0.4 cm delay between Trace A and Trace B represents a time delay of 0.04 sec. The velocity of wave is then

$$v = \frac{0.0416 \text{ ft}}{0.04 \text{ sec}} = 10.4 \text{ fps}$$

Figure 4

The Variation of Local Liquid Particle Concentration In the Vapor Core Center Versus Percent Liquid In the Total Flow

Test No.

70664-1 V_{vmax} = 722 **f.p.s.**
70864-1 V_{vmax} = 809 **f.p.s.**
71664-1 V_{vmax} = 876 **f.p.s.**
72064-1 V_{vmax} = 527 **f.p.s.**

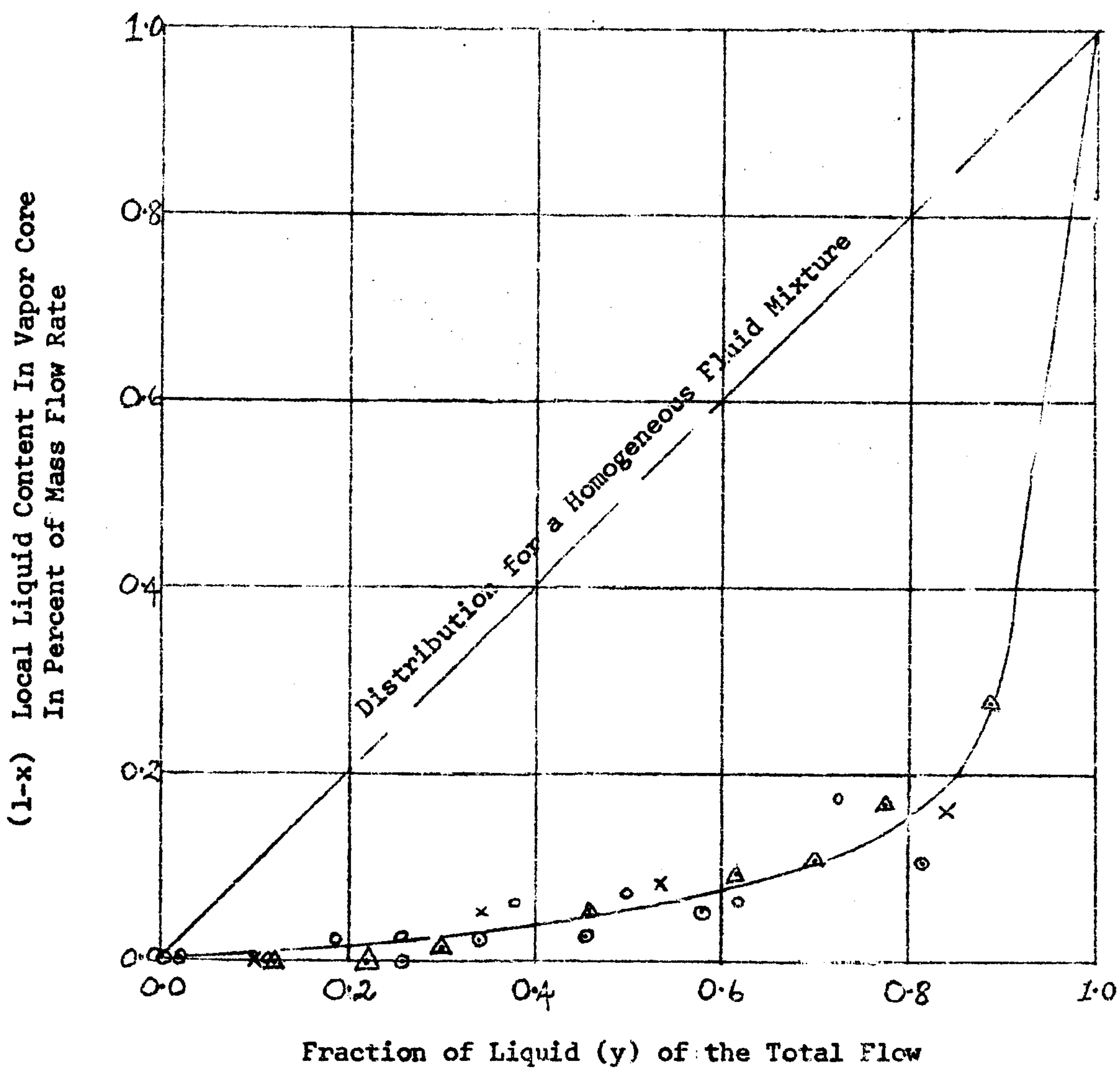

Figure 5

The Variation of Local Liquid Particle Concentration In the Vapor Core Center Versus Percent Liquid In the Total Flow

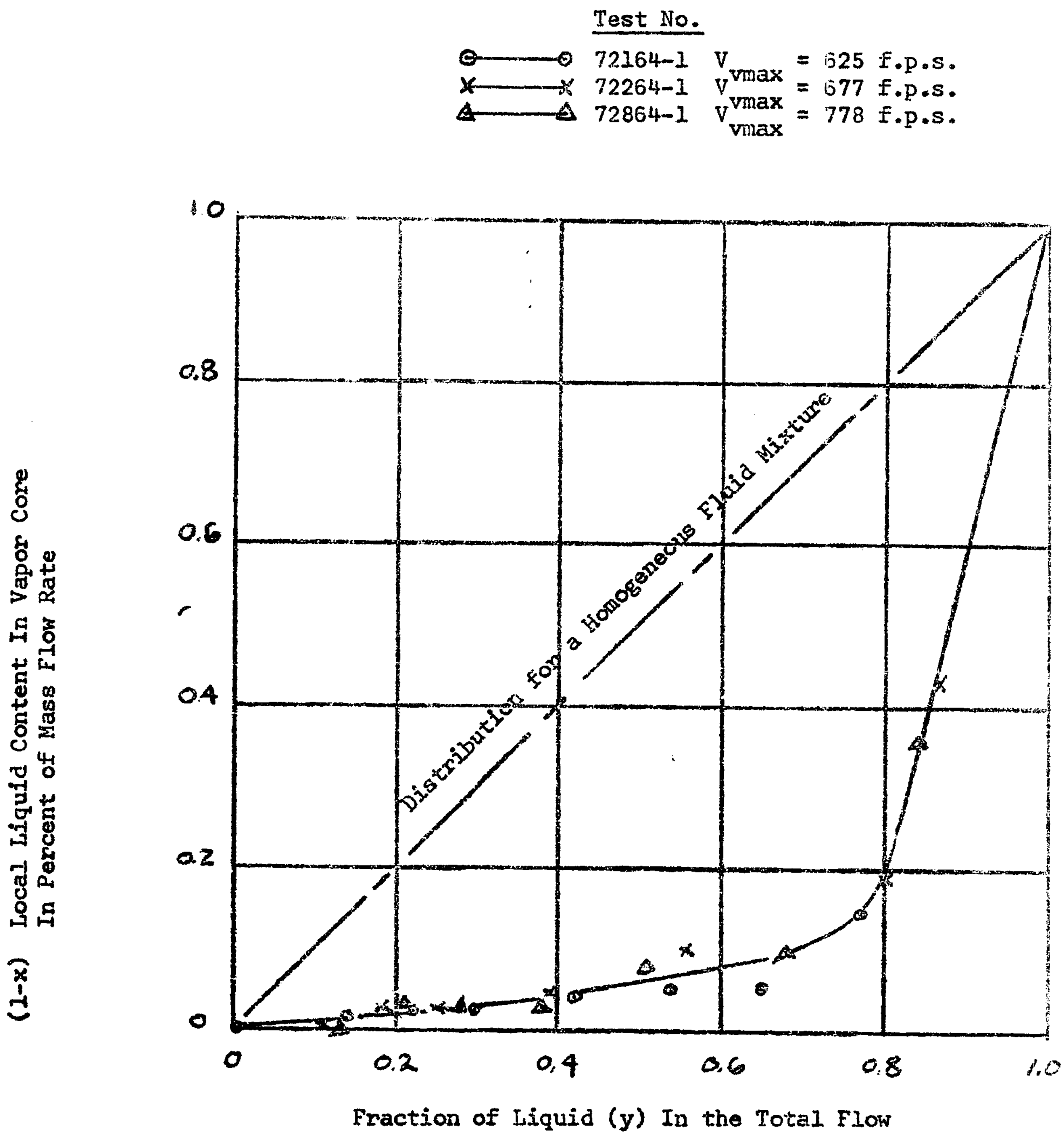

Figure 6

The Variation of Local Liquid Particle

Concentration In the Vapor Core Center

Versus Percent Liquid In the Total Flow

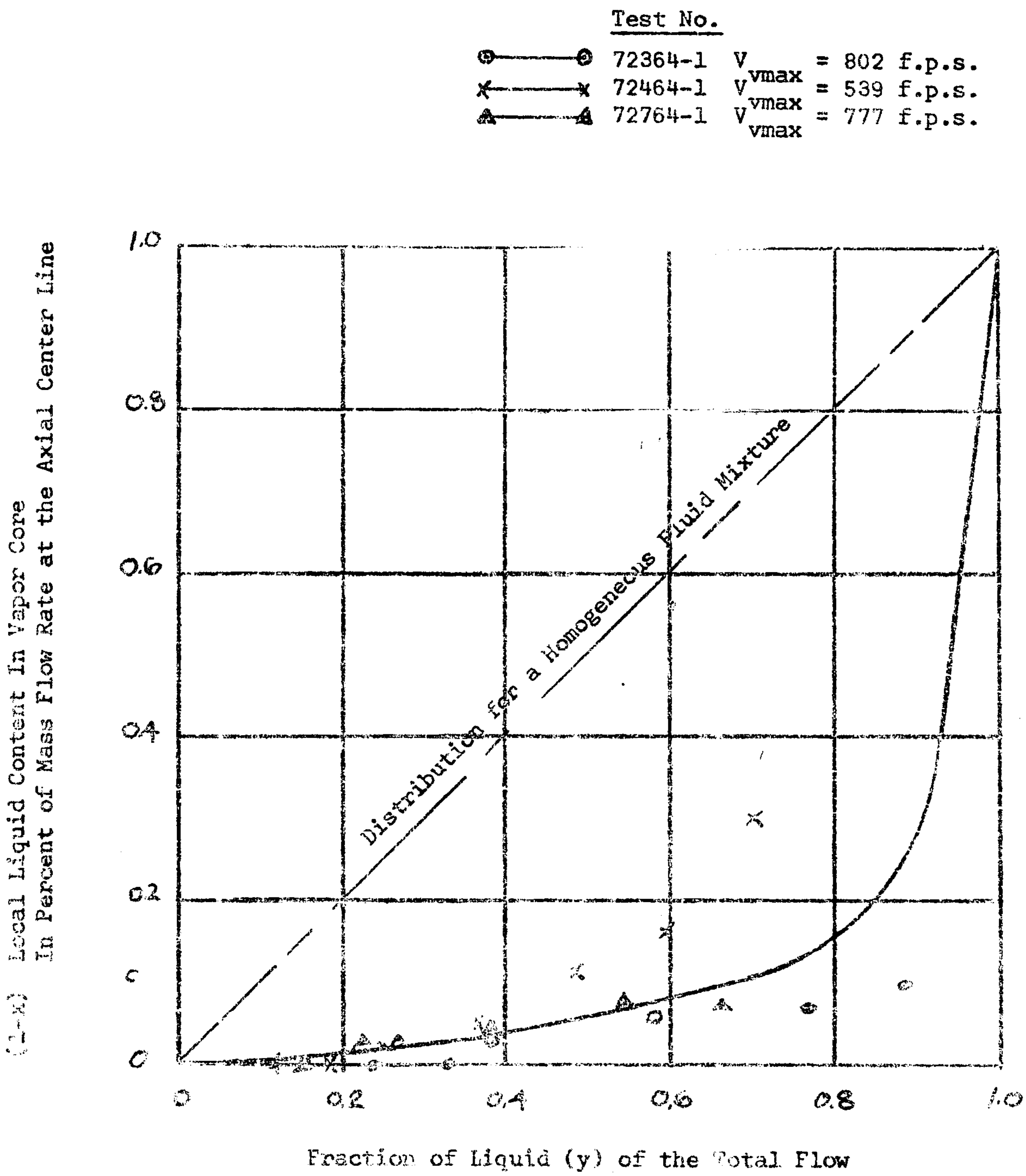

Figure 7

```
      DIMENSION TCW(21),PS(21),PSTV(21),VL(100),VV(21),Q(21),WV(21),
     1 AV(21),HSTV(21),TSTV(21),SSTV(21),TS(21),HL(21),DENL(21),
     2  DENV(21), AL(21), WL(21) ,HV(21),SPVOL(21),SPVOV(21),SV(21)
        REAL MT
      READ(5,1) MT,A,TEST,HO,W,TYPE,EPSLON
    1 FORMAT(2E12.4,I5,F7.2,F7.3,I2,F7.4)
        READ(5,2)((TCW(I),PS(I),PSTV(I)),I=1,21)
    2  FORMAT(10F7.3/10F7.3/10F7.3/10F7.3/10F7.3/10F7.3/3F7.3)
        WRITE(6,3)
    3   FORMAT(16H INPUT DATA ....)
      WRITE(6,4)
    4 FORMAT(77H  TEST  TOTAL MASS FLOW  TOTAL AREA  STAG. ENTHALPY  COO
     1LING WATER FLOW  TYPE   )
        WRITE(6,5)TEST,MT,A,HO,W,TYPE
    5  FORMAT(I7,F17.8,F12.6,F16.6,F20.9,I5)
        WRITE (6,6)
    6 FORMAT(59H  I COOL.WATER TEMP.     STATIC PRES.    STAGNATION PRES
     1S.    )
        WRITE(6,7)((I,TCW(I),PS(I),PSTV(I)),I=1,21)
    7  FORMAT(I5,F14.7,F18.7,F20.9)
    8 FORMAT(124H  I VAP VEL  HEAT OUT  VAP FLOW  LIQ VEL  LIQ FLOW  VAP
     1 AREA  LIQ AREA  LIQ ENTHPY  VAP ENTHPY  STAG ENTHPY  STAG TEMP  J
     2    )
         I=0.
   10    I=I+1
      CALL STEAMT(TS(I),PS(I),7,3,5,SPVOV(I),H)
      CALL STEAMT(TS(I),PA,4,1,S,SPVOL,HL(I))
      CALL STEAMT(TS(I),PA,6,1,S,SPVOL(I),H)
      CALL ISENT(1,Q1,PS(I),T1,HV(I),SV(I),I2,Q2,PSTV(I),TSTV(I),
     1 HSTV(I),SSTV(I),PT)
       VV(I)=SQRT(2.*32.2*778.*(HSTV(I)-HV(I)))
       Q(I)=W*(TCW(1)-TCW(I))
      J=1.
        VL(J)=0.
   15 WV(I)=(Q(I)-HO+MT*(HL(I)+VL(J)**2/(64.4*778.)))/((VL(J)**2-
     1VV(I)**2)/(64.4*778.)+HL(I)-HV(I))
         DENV(I) =1./SPVOV(I)
          AV(I)=WV(I)/(DENV(I)*VV(I))
        AL(I)=A-AV(I)
      WL(I)=MT-WV(I)
         DENL(I) =1./SPVOL(I)
      J=J+1
         VL(J)=WL(I)/(DENL(I)*AL(I))
      IF(ABS(VL(J)-VL(J-1))-EPSLON)25,25,20
   20 WRITE(6,21)J,VL(J)
   21 FORMAT(15H   J  LIQ VEL   /I4,F12.6)
   22 IF(J-100)15,40,40
   25   VL(I)=VL(J)
        WRITE (6,8)
      WRITE(6,30)I,VV(I),Q(I),WV(I),VL(I),WL(I),AV(I),AL(I),HL(I),
     1 HV(I),HSTV(I),TSTV(I),J
   30  FORMAT(I3,F9.4,F10.4,E10.4,F9.4,E10.4,E10.4,E10.4,F12.6,F12.6,
     1  F13.6,F10.5,I3)
         IF(I-21) 10,40,40
   40   CALL EXIT
         END
```

Figure 8

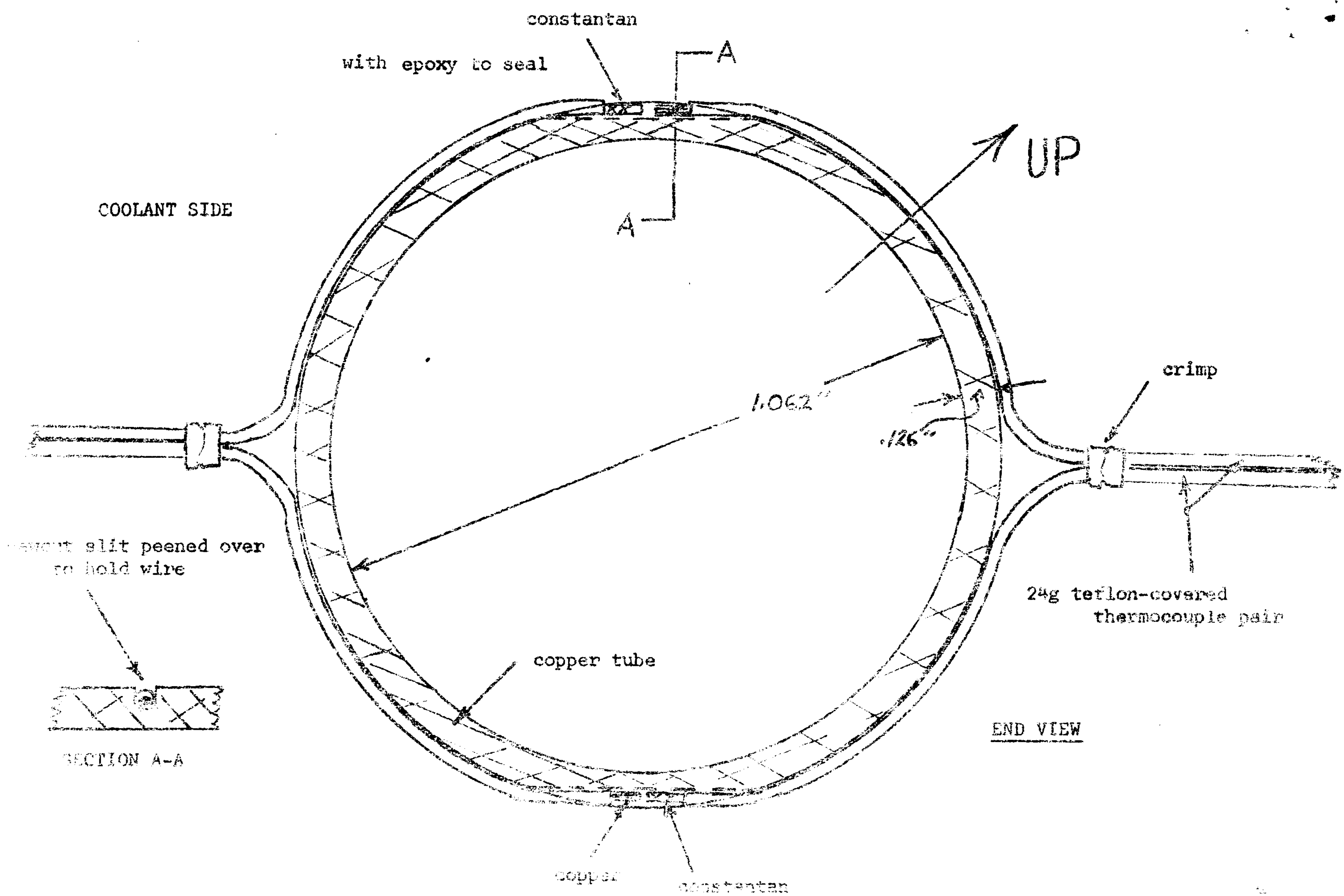

Figure 10 Schematic of New Wall Thermocouple Installation

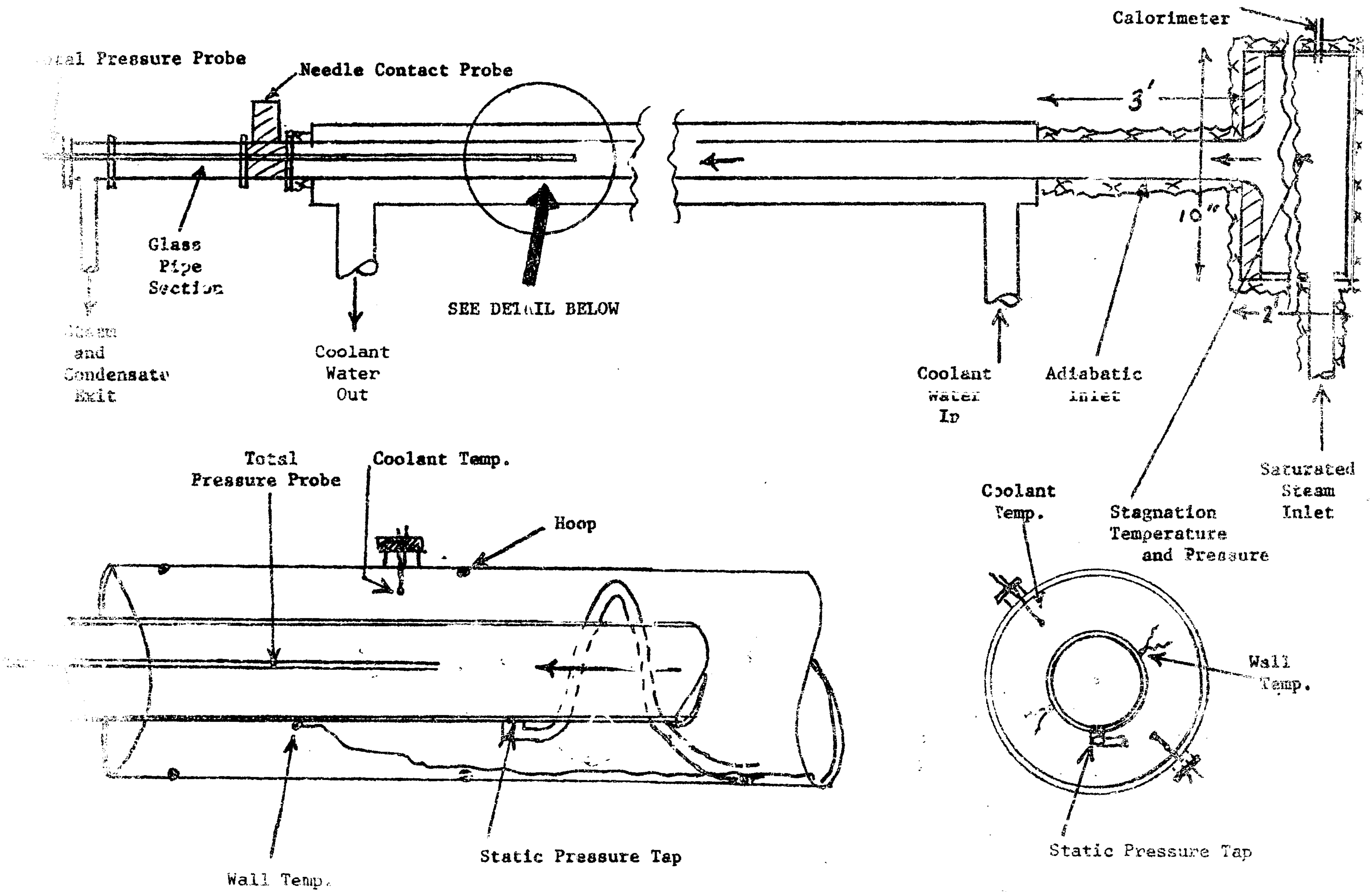

Figure 9

SCHEMATIC OF TEST SET-UP SHOWING PROPOSED CHANGES (Refer to Figures 1 and 2 of Progress Report, Nov. 1963)